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INDUSTRIAL APPLICATION

Suspension system performance optimization with discrete design variables

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Abstract Suspension systems on commercial vehicles have become an important feature meeting the requirements from costumers and legislation. The performance of the suspension system is often limited by available catalogue components. Additionally the suspension performance is restricted by the travel speed which highly influences the ride comfort. In this article a suspension system for an articulated dump truck is optimized in sense of reducing elapsed time for two specified duty cycles without violating a certain comfort threshold level. The comfort threshold level is here defined as a whole-body vibration level calculated by ISO 2631-1. A three-dimensional multibody dynamics simulation model is applied to evaluate the suspension performance. A non-gradient optimization routine is used to find the best possible combination of continuous and discrete design variables including the optimum operational speed without violating a set of side constraints. The result shows that the comfort level converges to the comfort threshold level. Thus it is shown that the operational speed and hence the operator input influences the ride comfort level. Three catalogue components are identified by the optimization routine together with a set of continuous design variables and two operational speeds one for each load case. Thus the work demonstrates handling of human factors in optimization of a mechanical system with discrete and continuous design variables.

Keywords Ride comfort · Whole-body vibration · Off-highway vehicles · Suspension · Hydraulic-mechanical system design · Discrete design variables

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1 Introduction

Suspension systems has become an important feature and competitive parameter in commercial vehicles and especially for off-highway vehicles travelling in rough terrain at relative high speeds such as dump trucks, forwarders and agricultural tractors.

There are several reasons for that. The European Union (2002) defined some limits for the daily exposure to whole-body vibration (WBV) experienced by the human operator and the manufacturers are according to the directive of the European Union (2006) required to declare if there are any occupational health risk associated with use of the machine, thus declaring the expected WBV exposure for a work day.

In Tüchsen et al. (2010) it is concluded that exposure to WBV predicts subsequent disability pension retirement for operators of commercial vehicles and off-highway mobile machinery. WBV is evaluated by the international standard 2631-1 by ISO (1997) and research carried out by Wikström et al. (1991) and Els (2005) shows a correlation between the measured WBV level and the subjective experience of ride comfort. WBV is probably the best measure to evaluate ride comfort in commercial vehicles and thus improving the level of WBV will also improve the comfort experience of ride.

Measurements in different fields indicates that transit mode often exposure the operator to the highest level of WBV compared to other operation modes (Tiemessen et al. 2007; Jönsson and Löfroth 2007; Griffin et al. 2006; Mayton 2009). Additionally increased travel speed will result in higher WBV exposure values which among others is shown in Hansson (1995), Uys et al. (2007) and Velmurugan et al. (2011). The speed dependency is in agreement with the experiences in the sales market; when the operator is equipped with a better suspended machine he will increase the travel speed typically up to a daily WBV exposure about 0.8 $\frac{\mathrm{m}}{\mathrm{s}^2}$ which tends to be the pain threshold.

This prompt a main challenge for the manufacturers of off-highway vehicles; Design and configure a suspension system that allows maximal travel speed without exceed the pain threshold of WBV of 0.8 $\frac{\mathrm{m}}{\mathrm{s}^2}$.

Improvement of suspension performance of the vehicle requires proper choices of the components for the suspension systems. Often the design will be restricted by commercially available components. Evaluation and improvement of ride comfort is carried out most cost efficiently by model based prototyping (Filla and Palmberg 2003). In a computer model it is possible to change parameters and evaluate the performance in a cost efficient way compared to changing and testing a full scale prototype. But the model has to be able to evaluate the ride comfort in terms of WBV. The calculation scheme for this is specified in ISO 2631-1. Doing the evaluation by simulation models requires a three dimensional multibody simulation model with the ability to handle off-highway soil conditions.

In this paper a hydraulic suspension design for a 10 tonnes articulated dump truck, Fig. 1, is presented and possible design variables are identified. Some design rules are introduced that effectively limits the feasible design space improving the chances of finding a proper design. Multibody dynamics simulation is applied together with a non-gradient optimization routine to find the best combination of components without violating the side constraints.

Fig. 1 Hydrema 912DS articulated dump truck

2 Modeling

A/S Hydrema Produktion has in collaboration with Aalborg University and University of Agder developed an in-house three dimensional multibody code in Fortran 90. The program is able to simulate some of the articulated wheeled Hydrema vehicles. The program is build up in modules and the model can easily be parameterized. For this research the multibody simulation code has been used to simulate the Hydrema 912DS Dumper, Fig. 2.

The governing equations for the mechanical system consist of a set of second order differential equations and a set of algebraic equations. The entire set of equations is listed in (1):

$$\begin{bmatrix} \mathsf{M} & -\Phi_q^T \\ \Phi_q & 0 \end{bmatrix} \begin{bmatrix} \dot{\mathbf{h}} \\ \lambda \end{bmatrix} = \begin{bmatrix} \mathbf{g}^{\text{ext}} - \mathbf{b} \\ \gamma \end{bmatrix} \tag{1}$$

The governing equations for the hydraulic systems consist of a set of first order differential equations given by:

$$\dot{\mathbf{p}} = \mathsf{C} \cdot \mathbf{h} + \pi \tag{2}$$

In (1) $\mathbf{g}^{\text{ext}}$ is a vector containing the generalized external forces acting on each body. Gravity, hydraulic cylinder forces, springs and dampers, propel torque and the tire-ground interactions all contribute to components of $\mathbf{g}^{\text{ext}}$. The stiffness and damping characteristics of rubber bushing and springs and dampers are described by linear and second order formulations. The wheels are modeled as individual bodies. The external force components acting on the wheels are the reaction forces between the terrain and wheels. The force contributions from the hydraulic cylinders are calculated from the pressure states in the hydraulic circuits such as stabilizing system, steering system and suspension system. The pressure gradients, $\dot{\mathbf{p}}$, are computed from (2) that is based on mass conservation and assumed Newtonian fluids yielding a set of decoupled equations.

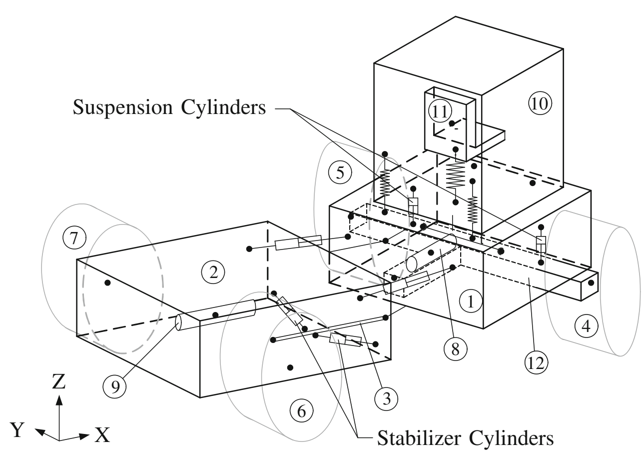

Fig. 2 Hydrema 912DS articulated dump truck consisting of 12 bodies

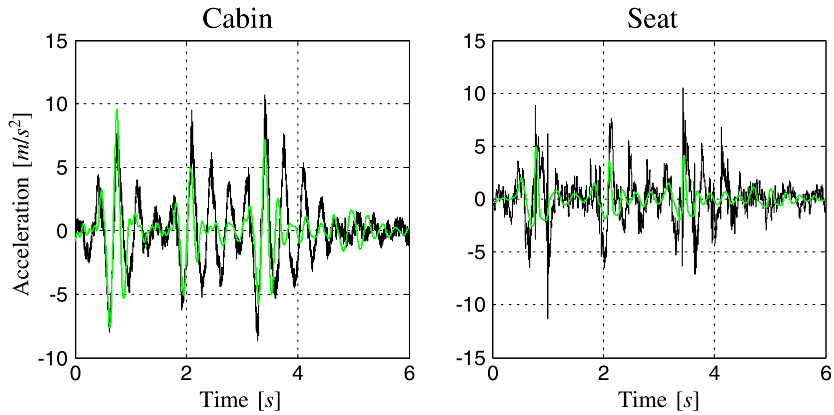

Fig. 3 Comparison of vertical accelerations from the simulation (*green*) of the cabin and seat with the dynamic response of full scale measurement (*black*)

For the numerical time integration of the initial value problem a fixed step integrator is applied. The dynamics of the hydraulic system is considerably stiffer than the dynamics of the mechanics. Therefore multirate integration is applied to handle the hydraulic states of (2) inspired by Buzdugan et al. (1999) and Gonzalez et al. (2009).

In order to be able to evaluate realistic ride comfort on off-highway conditions the tire response is crucial. In the multibody simulation code a tire model developed for off-highway vehicles with big tires is applied. The model needs only few modeling parameters and is able to handle even rough terrain with obstacles, see also Langer et al. (2009).

To control the dumper two inputs are used. The first one is the forward speed. A reference speed is given and a PI Controller adjusts the torque on the drive shaft and apply an opposite torque where the engine is mounted. The other input is a path that the dumper should follow. The articulated steering is handled by a double spherical joint, described in Nikravesh (1988, Chap. 7) with variable length simulating one of the steering cylinders in the articulated joint. The length is given by a controller that convert the offset between the dumper and the path to a reference length of the steering cylinder, described more detailed in Saeki (2002). The built-in constraint violation stabilizing method, described in Nikravesh (1988, Chap. 13) with both α and β equal to 10, then controls the joint forces needed to obtain the reference length.

In order to evaluate the WBV level in accordance with the ISO 2631-1 digital filters are build into the model to handle the frequency weightings of the acceleration response. The digital filters are implemented as described in Rimell and Mansfield (2007) and facilitate the calculation of the vibration exposure for each of the three directions as the RMS value of the frequency weighted acceleration:

$$a_{w(direc)} = \left[\frac{1}{T} \int_0^T a^2_{w(direc)}(t)dt \right]^{\frac{1}{2}} \tag{3}$$

Now the WBV level is defined as the largest of the three vibration values with a scalar weighting on the x and y direction according to ISO 2631-1:

$$a_{w,\max} = \max \left\{ 1.4a_{w,x};\, 1.4a_{w,y};\, a_{w,z} \right\} \tag{4}$$

Before any design changes was made the dynamic response of the existing dumper has been compared to measurement, Fig. 3, for a specific test track. The simulated response of the cabin and seat is in reasonable accordance with the reality. The vertical frequency weighted root-mean-square acceleration a_{wz} for the simulation model is 0.86. The measured a_{wz} is 1.11. By experience the measured value will always be higher than the simulated because of the contribution of vibration from other sources in the hydraulic and mechanical system. Even in idle mode a low level of WBV will occur. Though there is a difference between simulation and measurements it is concluded that the model is well suited for evaluating the performance since the frequency response (especially for the wheels according to Langer et al. 2009) fits quite well which is considered crucial when evaluating WBV in the low frequency range and adjusting damping parameters highly affected by the high frequency range.

3 Considered system

As the Hydrema 912DS on Fig. 2 has stiff axles the topology shown in Fig. 4 has been chosen as front axle suspension. The front axle is mounted at the tractor part behind the axle in a spherical joint. The sideway forces are primarily transmitted through a Panhard tension-compression bar. The vertical forces are transmitted via the two suspension cylinders as seen in Fig. 4.

On Fig. 5 a simplified diagram is shown of one hydraulic suspension circuit. The ring side of the cylinder is

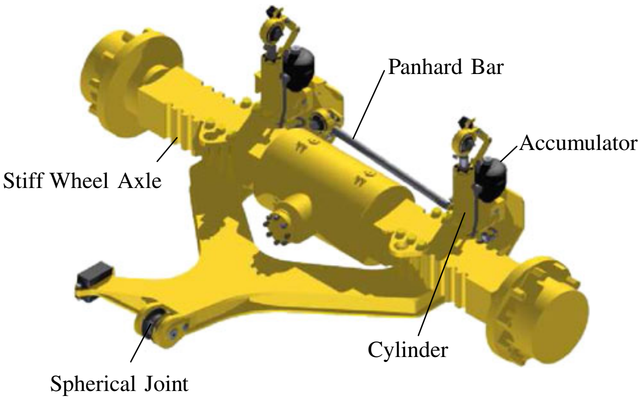

Fig. 4 Front axle suspension design

connected to tank pressure with a thick hose and big fittings to minimize flow induced pressure losses. The bottom side of the cylinder contains the oil pressure which carries the load from the dumper. This chamber is connected to a hydraulic accumulator through a direction valve parallel with an orifice.

Work of principal is that when the front wheel of the dumper hits an obstacle the oil from the bottom chamber should easily be displaced to the accumulator. The accumulator will afterwoods try to displace the oil back in the cylinder chamber when the pressure in the cylinder decreases again. On its way back is has to pass through the orifice.

Hereby the accumulator represents a hydro-pneumatic spring of a suspension system and the orifice represents damping. These represent the main components in the suspension circuit. Besides a very slow level adjustment system is connected to the suspension circuit. It can supply or displace oil from the circuits. Because the supply system is throttled down it has no influence of the dynamics and is not modeled.

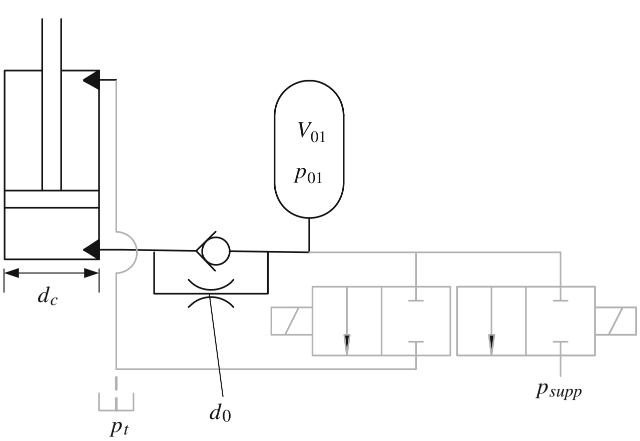

Fig. 5 Characteristics of direction valve and orifice

The direction valve is modeled as fully open/fully closed depending on the pressure difference between oil in the cylinder and the oil in the accumulator. The orifice is described by the orifice (5) assuming turbulent flow.

$$Q = C_d \cdot A_d \cdot \sqrt{\Delta p} \tag{5}$$

The accumulator is described by Rasmussen et al. (1996):

$$constant = p \cdot V^n \tag{6}$$

p is the pressure, V is the volume of the gas and n is the polytrophic exponent which can vary from 1 to 1.67 according to Giliomee (2005).

To find the operational contraction value C_d of the orifice (5) an experiment in Langer et al. (2010) have been carried out measuring the pressure difference between cylinder and accumulator at different cylinder travel velocities. Aware that the oil is compressible it still gives a reasonable determination of the contraction value C_d which is determined to approximate 0.95.

The dump truck on Fig. 2 is equipped with two stabilizing hydraulic cylinders between body 2 og 3 connected to a accumulator, Fig. 6. The cylinders are of special design only able to displace oil to the accumulator when longer than neutral length hence working as tension springs only. The hydraulic stabilizing system has a great influence on the roll stiffness between tractor and trailer part of the vehicle and hence the roll motion when turning.

The circuit is preloaded to a given pressure p_{stab} which together with the accumulator configuration and cylinder dimensions determine the roll stiffness.

The design space on the considered machine is limited. Hence the topology design changes are also limited. The stroke of the cylinders is given by the design specification to 0.08 m. Because of the limitation on the space, there are the following design variables left: volume of the suspension accumulator V_{01}, initial pressure of the suspension accumulator p_{01}, the diameter of the cylinder d_c, the diameter of the return orifice d_o, the volume of the stabilizer accumu-

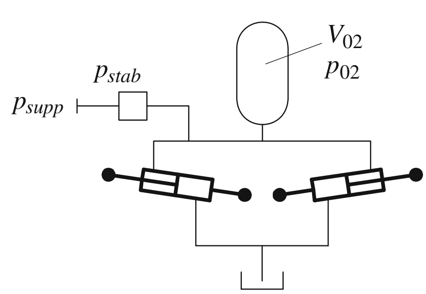

Fig. 6 Pendulum bar stabilizing system circuit

lator V_{02}, initial pressure of the stabilizer accumulator p_{02} and preload pressure in the stabilizer circuit p_{stab}.

Besides the system design variables the selected forward speed of the empty vehicle v_e and fully loaded vehicle v_f respectively will determine the suspension system performance.

Observing the nine design variables reveals that they may be divided into two categories:

1. Discrete: $\mathbf{y}^d = \{ V_{02} \ V_{02} \ d_c \}$
2. Continuous: $\mathbf{y}^c = \{ p_{01} \ p_{02} \ d_0 \ p_{\text{stab}} \ v_e \ v_f \}$

Hydraulic cylinders are produced by Hydrema in standard dimensions. Some possible cylinders are listed in Table 2 in Appendix. From the supplier a number of nitrogen accumulators are extracted in a range of feasible sizes, Table 3 in Appendix. Each accumulator are subjected to some properties in terms of the permitted operating pressure p_{per} it can stand and a permitted pressure ratio PPR which is the maximum between the duty pressure and the initial pressure of the accumulator p_0.

4 Improvement of performance

An optimization problem is in general formulated as the minimization of an objective function O subjected to n inequality constraints and m equality constraints:

$$\min \ O(\mathbf{y}) \tag{7}$$

$$g_i(\mathbf{y}) < 0.0 \qquad i = 1 \ldots n \tag{8}$$

$$h_i(\mathbf{y}) = 0.0 \qquad i = 1 \ldots m \tag{9}$$

where, $\mathbf{y} = \begin{bmatrix} \mathbf{y}^{(d)} \\ \mathbf{y}^{(c)} \end{bmatrix}$, is the vector of design variables. The objective function and the inequality constraints may be combined to form an augmented penalty function $\Theta(\mathbf{y})$ that can be subjected to minimization:

$$\min \Theta(\mathbf{y}) = O(\mathbf{y}) + \sum_{i=1}^{n} G_i(\mathbf{y}) \tag{10}$$

$$G_i(\mathbf{y}) = \begin{cases} 0 & g_i(\mathbf{y}) \geq 0 \\ g_i^2(\mathbf{y}) & g_i(\mathbf{y}) < 0 \end{cases} \tag{11}$$

In this case the objective function $O(y)$ is the elapsed time for some defined tasks.

For agricultural tractors a standardized test track, ISO 5008, is used to evaluate the ride comfort (ISO 2002). To evaluate the suspension system performance of the Hydrema dump truck the smooth track specified in ISO 5008 is applied, Fig. 7.

An articulated vehicle is to a high degree characterized by the displacement of the center of gravity when the vehicle is steered out. This results in an unsymmetrical load distribution on the wheels, Fig. 8, and some stability issues often have to be considered (Dudzinski 2005, Chap. 6). Applying front axle suspension to an articulated machine will cause the tractor to body roll because of the asymmetric load distribution. The hydraulic stabilizing system between the tractor and trailer counteract for some of the body roll movement though. It is experienced that body roll movement of the tractor is subjectively discomfortable when the suspension cylinder stroke difference (SCSD) is over 0.03 m when fully steered out, which require a side constraint to the optimization of the suspension configuration. This is evaluated by a short simulation with and without payload making a fully right turn.

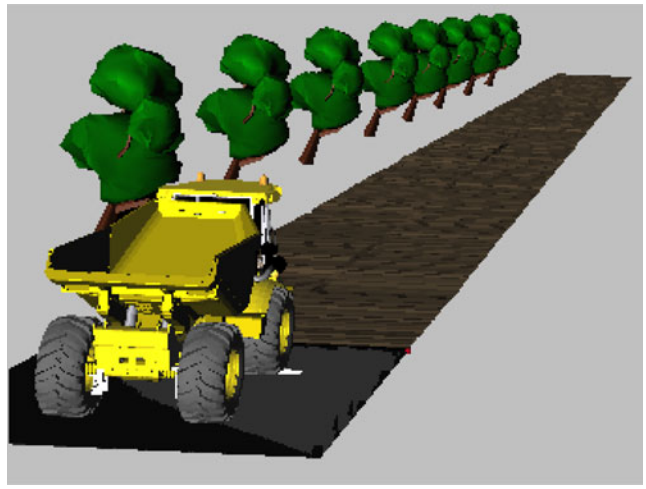

Fig. 7 ISO 5008 smooth track

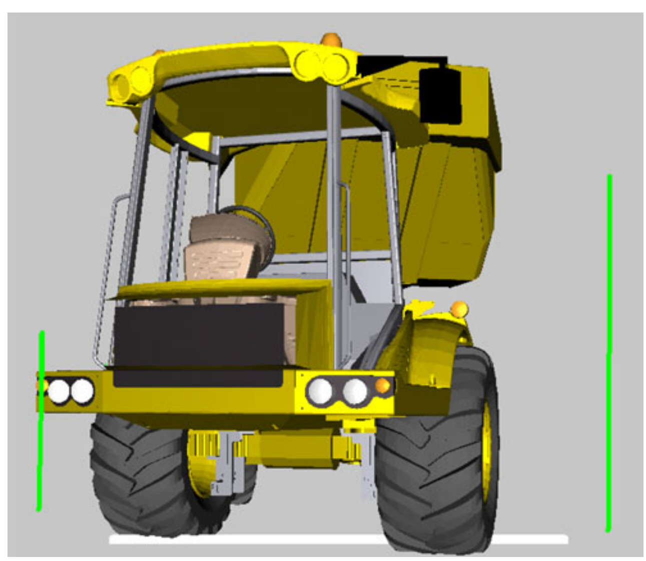

Fig. 8 Load distribution at the front wheels when fully steered out

Hereby four simulations are used to evaluate the quality of a configured suspension:

1. Fully right turn without payload.
2. Fully right turn with payload.
3. 10-m travel without payload at the ISO 5008 smooth track.
4. 10-m travel with payload at the ISO 5008 smooth track.

Besides the side constraint of the SCSD, there are 12 other side constraints for the suspension system given by the chosen design and the supplier of the accumulators (Hydac 2011):

1. The summarized WBV level from (4) of the two travel modes should not exceed 0.8 $\frac{\mathrm{m}}{\mathrm{s}^2}$.
2. Maximum peak pressure should not exceed $p_{01} \cdot PPR$ for the suspension accumulator.
3. Maximum peak pressure should not exceed $p_{02} \cdot PPR$ for the stabilizer accumulator.
4. Maximum peak pressure should not exceed p_{per1} for the suspension accumulator.
5. Maximum peak pressure should not exceed p_{per2} for the stabilizer accumulator.
6. The gas volume should not exceed $V_{01} \cdot 0.9$ for the suspension accumulator.
7. The gas volume should not exceed $V_{02} \cdot 0.9$ for the stabilizer accumulator.
8. The gas volume should not be less than zero for the suspension accumulator.
9. The gas volume should not be less than zero for the stabilizer accumulator.
10. The suspension cylinders should not reach the end stop in the bottom.
11. The suspension cylinders should not reach the end stop in the top.
12. The minimum peak pressure should not be less than zero in the suspension system to avoid cavitation.

The contributions from the side constraint violations to the augmented objective functions are all normalized. The value of the augmented objective function appears by adding the total elapsed time of the two 10-m run with the square of all side constraint violations. According to Toscano (2009) no longer track is needed for comfort evaluation.

Hansen and Andersen (2005) suggest that the discrete design variables are handled as continuous and then, in order to obtain an optimal design that is actually useful to apply on real world applications, an integer evaluation number G_u is added to the augmented objective function:

$$G_u = \sum_{j=1}^{2} \left(\frac{u_i - i}{\epsilon} \right)^2 \tag{12}$$

where i is the rounded integer value of u and ϵ is a parameter that is gradually reduced. In practice this is done by listing the discrete design variables in tables like Tables 2 and 3 considering the index number at the left as the design variables and then interpolating between the rows when doing the evaluation.

To minimize $\Theta(\mathbf{y})$ the complex method first presented by Box (1965) is applied. It is a non-gradient based mapping method that uses a population of design called a complex. The complex contains q design. Each configuration is evaluated with respect to (10). Then the design with the highest objective value is changed. This is done by mirroring it in the center of the remaining design by:

$$\mathbf{y}_{\text{new}} = \kappa \cdot (\mathbf{y}_c - \mathbf{y}_{\text{worst}}) + \mathbf{y}_c, \mathbf{y}_c = \frac{\sum_{i \neq \text{worst}}^{q} \mathbf{y}_i}{q-1} \tag{13}$$

κ is called the reflection constant which is set to 1.3 as originally suggested by Box (1965). If the newly mirrored design continues to evaluate as the worst design it is moved toward the currently best design. The algorithm for this depends on the number of successive mirror operations that has returned the particular design as the worst design, k:

$$\mathbf{y}_{\text{new}} = \frac{1}{2} \cdot (\mathbf{y}_{\text{worst}} + \epsilon \cdot \mathbf{y}_c + (1-\epsilon) \cdot \mathbf{y}_{\text{best}}) + \sigma \tag{14}$$

$\mathbf{y}_{\text{best}}$ is the currently best design and ϵ is:

$$\epsilon = \left(\frac{n_r}{n_r + k - 1} \right)^{\left(\frac{n_r + k - 1}{n_r} \right)} \tag{15}$$

In (15) n_r is a tuning parameter set to 4 and σ in (14) is an extra term suggested by Andersson (2000) that takes into account a situation where the centroid $\mathbf{y}_c$ coincides with a local minimum of the objective function:

$$\sigma = (\mathbf{y}_c - \mathbf{y}_{\text{best}})\,(1-\epsilon)(2 \cdot R - 1) \tag{16}$$

R is a random number between 0 and 1.

Box (1965) suggests a population size q that is twice the number of design variables and that is adopted here with a population q of 22.

5 Results

In Figs. 9 and 10 the development in the augmented objective function $\Theta(\mathbf{y})$ for the fittest member of the population is shown in the two intervals; 1–280 and 281–474 iterations, respectively. The criteria of converge is that the difference between the best and worst configuration should be less than 0.01. When ϵ is reduced, an increase of the fittest member is observed, and then the solution converge again but to a higher value than before, observed at Fig. 10.

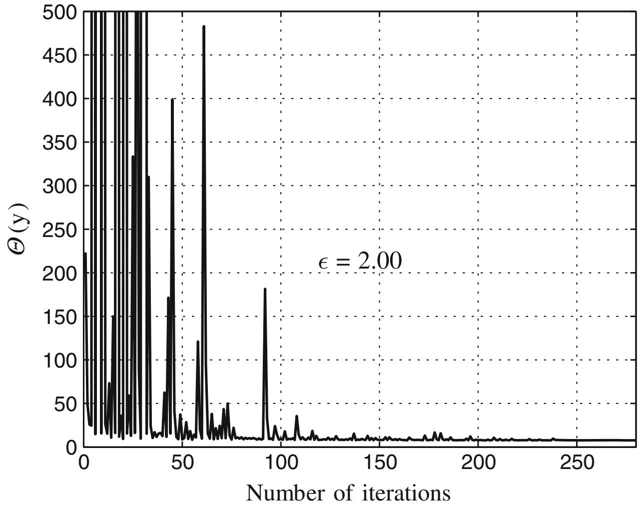

Fig. 9 Development of augmented objective function value of fittest member with $\epsilon = 2.0$

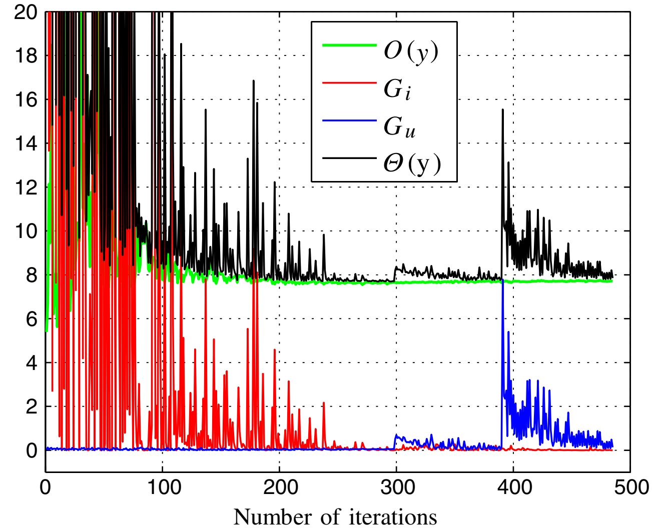

Fig. 11 Development of the objective function $O(\mathbf{y})$, side constraints violation G_i, integer evaluation G_u and the augmented objective function $\Theta(\mathbf{y})$

Figure 11 shows the total number of the iterations for $\Theta(\mathbf{y})$, which is the sum of the objective function $O(\mathbf{y})$, the side constraints violation G_i and the integer evaluation G_u. It is observed that the integer evaluation does not affect the evaluation for the first 295 iterations. Instead the side constraints govern the optimization process in the beginning. From iteration 295 the side constraints are not violated and the objective $O(\mathbf{y})$ has converged. Looking very carefully at Fig. 11 it is observed that the elapsed time increased slightly compromising integer values of component selection.

In Fig. 12 the evaluation value of WBV is shown. It is observed that the development of the evaluated WBV value is approaching approximately 0.8 $\frac{\mathrm{m}}{\mathrm{s}^2}$, which is the value of the pain threshold introduced by the side constraint.

In Fig. 13 the development of the cylinder diameter and the accumulator sizes are shown by table indices. The values moves toward integer values after iteration number 295 when the ϵ is decreased.

The design variables moves towards the nearest integer value in Fig. 13. This could be caused by a complex consisting of a very narrow sphere. With other words the solution have converged so much, that the complex is not able so seek outside a sphere and thus unable to find other combinations. To investigate this an optimization has been carried

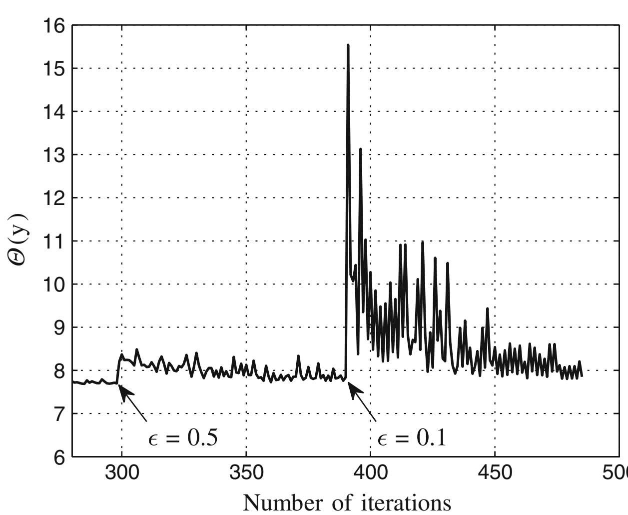

Fig. 10 Development of augmented objective function value of fittest member with decreased ϵ

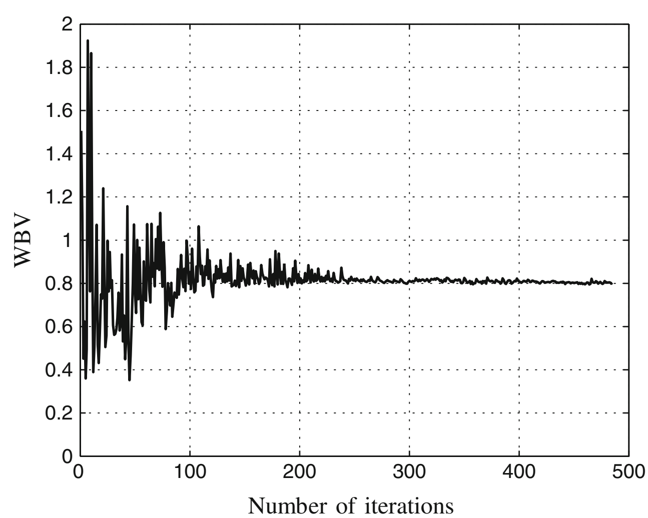

Fig. 12 Development of the WBV value acting as a side constraint

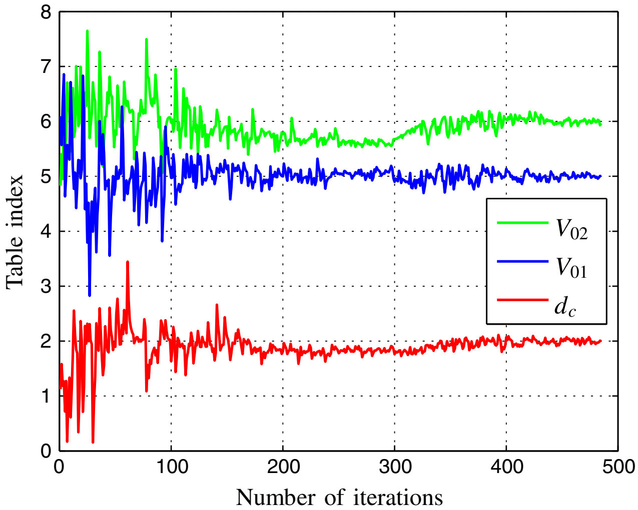

Fig. 13 Development of the three discrete design variables with decreasing ϵ

out with $\epsilon = 0.1$ from the start and with no changes through the optimization. The development of the discrete design variables follows in Fig. 14 resulting in the same component combination.

In Table 1 the optimized design of the coupled stabilizing and suspension system is shown. This design gives the highest possible performance in combination with the selected forward speeds $v_e = 3.05 \frac{\text{m}}{\text{s}}$ and $v_f = 2.31 \frac{\text{m}}{\text{s}}$ for empty and fully loaded vehicle respectively. The final design combination is about three times as good as the best member of the start population.

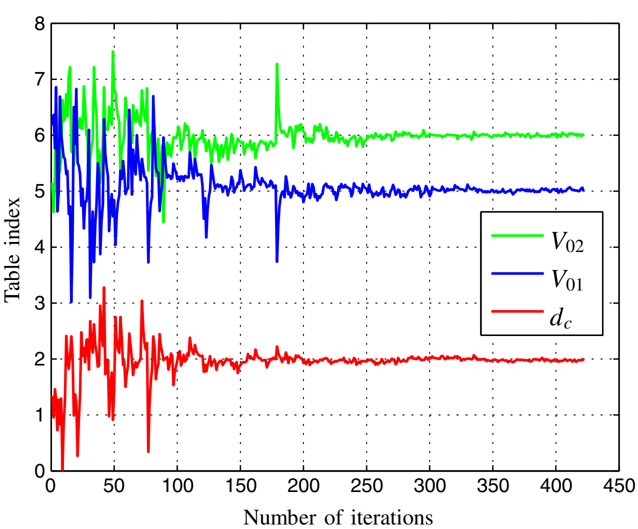

Fig. 14 Development of the three discrete design variables with constant ϵ

Table 1 Final optimized design

Design variable	Value
V_{01}	0.60 l
p_{01}	16 bar
d_c	0.080 m
d_0	0.00323 m
V_{02}	0.75 l
p_{02}	33 bar
p_{stab}	49 bar

The polytrophic exponent n can vary in a quite wide range (Giliomee 2005). Though in Langer et al. (2010) it is tested that the polytrophic exponent does not affect the WBV evaluation considerable in simulations with the described modelling technique.

6 Conclusions

Suspension systems is important to gain market advantages for manufacturers of commercial vehicles since the performance of the vehicle will among others be constrained by the pain threshold of the operator in terms of WBV due to dynamic response of the vehicle.

Improving suspension will improve both comfort and productivity of the vehicle depending on the operation conditions. And at the same time improve the possibility for the operator to meet the legal requirements regarding to vibration exposure.

By a non-gradient optimization method the optimum configuration of seven design variables together with appropriate speeds are fund for a hydro-pneumatic suspension system for an articulated dump truck. The evaluation criterion is a minimum elapsed time for two transit modes with a number of side constraints. Among these a maximum level of WBV of 0.8 $\frac{\text{m}}{\text{s}^2}$, which restrict the design variables and the forward speed. The results show that WBV level indeed is dependant of travel speed. The simulation time for 500 iterations is about 12 h, which is within an acceptable simulation time equal to one night.

This is the first attempt to optimize a model based suspension system on the Hydrema 912 dump truck. The cylinder travel is highly restricted due to acompact and narrow geometry of the chassis. The initial guess of the design variables are based on the existing stabilizing system, a suspension cylinder size that gives feasible circuit pressure (that is lower than the supply pressure of the machine in order to be able to raise a fully loaded machine) and an accumulator that exceeds the displaced volume of the suspension cylinder. Due to the limited suspension cylinder travel it is a challenge to find a feasible design that does not violate the side constraints. In Fig. 11 it is observed that the violation of the side constraints G_i drives the optimization process for the first 250 iterations. Hence, the accomplished optimization is more a way to identify a feasible design more than fine tuning the system. From iteration number 250 to 300 a feasible working design is fine-tuned, but the improvement in these 50 steps is a few percent only.

Three discrete design variables are handled as continuously and then an augmented objective restrict the design variables to become exact components in a library. Hereby it is demonstrated that discrete variables in design optimization of real world application can be done effectively. It is also shown that a carefully selected restriction value ϵ from the beginning can be used through-out the optimization without changing the value. For this system it is observed from Figs. 13 and 14 that handling the discrete design variables as continuous and picking the closest option will return the same combination of components as evaluation G_u. For most applications this will be the case. But applying integer evaluation with decreasing ϵ also tells the designer how much the limitation on available catalogue components compromises the performance of the system. Hereby, the designer can make an objective decision if one should usecatalogue components or require special designed components from ones one company or OEMs.

Compared to the unsuspended Hydrema 912 the WBV exposure with the optimized suspension design has reduced to the half when driving at ISO 5008 smooth track with the same reference velocity (Langer et al. 2010). Also for the empty machine the forward velocity v_e is identified to 3.05 m/s without violating the WBV side constraint. This is very close to the specified test speed of the smooth track of ISO 5008 which is 3.33 m/s, which is quite good (ISO 2002).

Hence the article demonstrate performance optimization of a suspension system taking the human limitation into account and at the same time handle discrete design variables to obtain an optimum combination of commercial available components. In the future it will be interesting to implement a semi-active control of the suspension cylinders to improve the ride-comfort. A simple way will be to control the orifice diameter d_0 in Fig. 5 (Savaresi et al. 2011).

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Appendix

This appendix contains table for standard sizes of hydraulic cylinders and hydraulic accumulators.

Table 2 Standard piston sizes

d_c	Piston diameter [m]
1	0.063
2	0.080
3	0.085
4	0.090
5	0.120

Table 3 Available accumulators sizes (Hydac 2011)

V_0	Volume V_0 [L]	Permitted operating pressure p_{per} [Bar]	Permitted pressure ratio PPR
1	0.075	250	8
2	0.16	300	8
3	0.32	300	8
4	0.50	210	8
5	0.60	330	8
6	0.75	330	8
7	1.00	200	8
8	1.40	330	8
9	2.00	330	8

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